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Parametric and exergetic analysis of waste heat recovery system based on thermoelectric generator and organic rankine cycle utilizing R123

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ABSTRACT

The paper analyzes the combined TEG-ORC (thermoelectric generator and organic rankine cycle) used in exhaust heat recovery of ICE (internal combustion engine) theoretically. A theoretical model is proposed to calculate the optimal parameters of the bottoming cycle based on thermodynamic theory when net output power and volumetric expansion ratio are selected as objective functions, which affect system performance and size. The effects of relative TEG flow direction, TEG scale, highest temperature, condensation temperature, evaporator pressure and efficiency of IHE (internal heat exchanger) on system performance are investigated. R123 is chosen among the fluids whose decomposition temperature exceeds 600 K to avoid fluid resolving and resulting in wet stroke when expansion process ends. The thermodynamic irreversibility that occurs in evaporator, turbine, IHE, condenser, pump and TEG is revealed at target working areas. The results indicate a significant increase of system performance when TEG and IHE are combined with ORC bottoming cycle. It is also suggested that TEG-ORC system is suitable to recovering waste heat from engines, because TEG can extend the temperature range of heat source and thereby improve the security and fuel economy of engines.

1. Introduction

With the rapid development of population and vehicle industry in the world during the past 20th century, the demand on passenger vehicles has increased sharply. Recent studies [1,2] indicate that, only 41% of a class-8 truck diesel engine's fuel combustion energy is converted into useful work to drive a vehicle and its accessory loads. The remainder is engine waste heat dissipated by engine exhaust system (20%), coolant system (18%), EGR (exhaust gas recirculation) cooler (11%), CAC (charge air cooler) (9%), and convection as well as radiation from engine block. This increases fuel consumption, which rises almost exponentially due to low engine efficiency and increasing demand on vehicles, brings serious energy crisis and has environmental effects. If this waste heat of engines could be recaptured efficiently, engine output power will be significantly enhanced without additional fuel consumption. Thus, large amount of fossil fuel can be saved [3,4] and much less harmful exhaust gases are dissipated to ambient environment when the same output power is generated [5,6]. Furthermore, global warming will be relieved. However, we should notice that the potential energy savings from improved energy efficiency are estimated using basic physics principles and engineering models. The actual energy savings from such improvements generally falls short of such estimates due to the rebound effect [7]. A possible explanation for this phenomenon could be that such improvements encourage the consumption of energy services where part or all of the gain would be offset by the increase in consumption [8]. Since the price of fossil fuel rises greatly due to serious energy crisis, renewable clean energy will play an important role in the future. However, with plenty of current technical obstacles, renewable technology cannot be widely applied to vehicle industry within short term. In other words, it is a considerable solution to improve engine efficiency via waste heat recovery system [3]. In particular, it has considerable potential environmental and economic benefits. Thus, many researches and projects focus on enhancing engine performance and thermal efficiency, aiming at lowering fuel requirement and exhaust that are harmful to human body. Converting exhaust heat to electricity by ORC (organic rankine cycle) is an interesting and actual avenue among the ways of recovering the waste heat [5,9].

Explosive development of ORC technology has been achieved in the area of WHR (waste heat recovery) of low temperature/grade heat sources, such as geothermal sources [10–12], solar energy [13–16], bio-fuel electricity production plants [17–20] and vehicle

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exhaust gases [1,3,5,21] during the last one hundred years. When it comes to engines, some obstacles should be cleared away before ORC technology can be applied to vehicle engines. Compared with the other two sources of low temperature, the temperature of IC engine exhaust gas is much higher, even than the decomposition temperature of most organic fluids, which generally are kept below 623 K. Thus, when organic rankine cycle technology is applied to IC engines, temperature difference between exhaust gas and organic working fluid becomes much greater. Working fluid may resolve due to high exhaust temperature, and then prevent WHR system from working fluently and safely. Previous studies [22,23] indicate that using water only as working medium for steam rankine cycle is one of the potential solutions of overcoming high exhaust temperature. Since decomposition temperature of water is up to 2273 K, this could avoid working fluid resolving but lead to a lower thermal efficiency and large system size and weight. In the 1970s, a steam rankine cycle system is applied to a 288-horsepower truck engine to recapture energy from exhaust gases by the Mack Trucks company. The actual bench test indicates that an increase of substantial fuel efficiency has been achieved [22]. However, it produces much lower power output and the devices are huge and heavy, so it is of less practical value. Thus, steam rankine cycle is not suitable to gain higher thermal efficiency, although it provides security for vehicle engines. Another solution is to lower exhaust gas temperature to guarantee working fluid work properly with high temperature exhaust gas, and this would waste part of total energy to make it adaptable to ORC system when exchanging heat with high temperature gases. Karellas et al. [10] proposed the case that exhaust exchanged heat firstly with thermal oil, and then the heated oil conducted heat to organic fluids to provide heat supply for ORC system. They also simulated and analyzed of ORC system with oil circulation to select the best refrigerant to drive thermodynamic cycle. They focused on system parameters optimization and redistributed the energy in the system. However, the studies show that these cases mentioned above cannot take full advantage of the high-grade waste heat of exhaust gas. Since the former case has low efficiency but needs large devices, it cannot be widely used in actual vehicle applications; and exhaust temperature of the latter case is much lower and a large portion of the exhaust energy is dissipated to ambient environment. It makes output power generated in bottoming cycle be greatly reduced and thermal efficiency get much lower, while oil circulation needs a secondary heat exchanger, which increases irreversible loss.

TEG (thermoelectric generator) has become another emphasis in the area of waste heat recovery in recent years [24–26]. But there hasn't been an abundance of literature and researches about using TEG to recover waste heat from automotive applications. That's because the conversion efficiency is low and the cost of thermoelectric material is high. But as the price of fossil fuel goes higher, and due to recent advancements in nanotechnology and semiconductor physics, more dedicated companies and government agencies contribute to improving and implementing the thermoelectric technology. As thermoelectric generator directly converts heat into electricity without moving parts and environmental side effects, and much less components are needed, thus less packaging and weight constraint compared with rankine cycle system, TEG method has become a renewed emphasis on vehicle applications in recent years with current nanotechnology applied in this area [27]. And it is interesting that Miller [28] tried to combine both TEG and ORC to recover heat from engines. Further researches [29,30] indicate a bottoming system combining TEG and ORC has potential for the area of waste heat recovery from engine exhausts.

There are a lot of difficulties in waste heat recovery from passenger vehicle engines. The most important one is that, high exhaust temperature would lead to great difference between refrigerant and exhaust gases. However, the maximum temperature of organic working fluid is much lower and would resolve if the temperature goes higher than decomposition temperature. Thus it's difficult to combine traditional ORC systems with heavy-duty vehicle engines and make full use of exhaust heat. Therefore, a system utilizing both TEG and ORC technologies, which would be called *TEG-ORC* System in the later sections, has been adopted to recapture power from exhaust gases in this paper. The working fluid won't resolve and additional power could be recovered, thus it effectively broadens the range of applications for ORC systems. In the bottoming cycle, the hot side of TEG would exchange heat directly with the original high temperature exhaust gases coming out from supercharger, while TEG releases a large portion of heat absorbed from exhaust into the working fluid at the cold joint, so the temperature of the working fluid becomes higher. Then the exhaust with lower temperature supplies the working fluid in the evaporator with adequate heat during evaporation and superheat processes. Besides, the optimization of the ORC working parameters is concluded in this paper. The combined TEG-ORC system is established on a diesel engine, and the computational model of analysis is developed using MATLAB/SIMULINK. The performance of the system has been simulated and analyzed to offer data supply for further test researches and vehicle applications.

2. System description

2.1. The topping ICE system

The diesel cycle of a commercial engine is considered as topping cycle [5]. The engine is an inline 6-cylinder 4-stroke supercharged diesel engine, and the main parameters of the engine are presented in Table 1. As the aim of the analysis in the study is to obtain parameters optimization for ORC use in exhaust heat recovery of ICE, we assume the engine to operate under rated conditions. It has been calculated that air fuel ratio is 28.59 and excess air coefficient is 2 under nominal conditions under the hypothesis of complete combustion of diesel oil. The composition of exhaust gases on mass basis has been calculated at: CO_2 = 10.78%, H_2O = 3.83%, N_2 = 74.16%, O_2 = 11.23%. This composition is used to evaluate gas properties. Energy analysis of WD10D235 engine without WHR system indicates that about 41% (235.8 kW power) of the total energy released from fuel combustion can be converted to useful work under rated conditions.

2.2. The bottoming TEG-ORC system

To enhance engine output power and reduce fuel consumption by recovering heat energy from engine exhaust gases, a novel waste heat recovery system has been established on WD10D235 engine in this paper. The bottoming system mainly consists of the following components (as shown in Fig. 1):

Table 1
Main parameters of the commercial WD10D235 engine.

Parameter	Value	Parameter	Value
Rated output power (kW)	235.8	Rotate speed (r/min)	1501
Torque (N m)	1500	Fuel consumption (kg/h)	47.79
Smoke intensity (FSN)	0.55	Combustion air mass flow (kg/h)	943
Exhaust temp. (K)	792.2	Exhaust mass flow (kg/s)	0.275
Coolant temp. (K)	356.5	Coolant mass flow (kg/s)	2.717

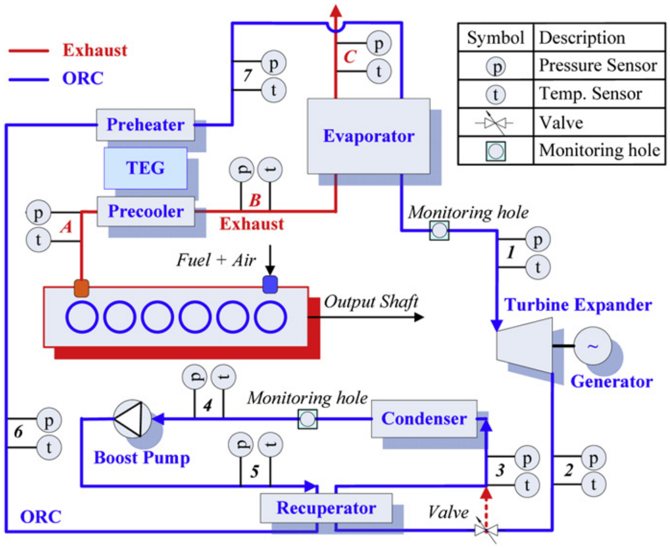


Fig. 1. Diagram of waste heat recovery system mounted on exhaust pipe of a diesel engine.

(1) Turbine Expander
(2) Generator
(3) Internal Heat Exchanger (IHE)
(4) Condenser
(5) Boost Pump
(6) Thermo-Electric Generator (TEG)
(7) Evaporator

A brief description of bottoming system operation is given below. The subcooled working fluid coming out from condenser is pumped to boiler pressure p_{evap} by boost pump. The cold liquid with high pressure then goes into recuperator and exchanges heat with expander exhaust with higher temperature. In the next section it works as the heat sink of TEG and its temperature increases. When the preheated working fluid goes into evaporator, phase change happens. The liquid temperature at the evaporator inlet (point 7) is determined by the performance of TEG and the efficiency of recuperator. Considering the changes of designed boiling pressure p_{evap} and condensation pressure p_{cond}, the vapor exhausted from the expander may result in wet saturated, dry saturated and superheated states, which rely on the property of different types of organic fluid. It is extremely harmful to the expander if exhausted vapor is in wet state. In order to avoid this wet stroke in expander, superheating of organic vapor at the inlet is necessary. The degree of superheating should be selected properly when different working fluids and working pressures are proposed for this system. Then the superheated vapor with high pressure and high speed works in the expander, where the kinetic and pressure energy converts into mechanical energy, and the expander drives the generator to produce electricity. The saturated or superheated exhaust vapor with low pressure goes into the recuperator. The vapor cools down in the internal heat exchanger by transferring heat to the compressed liquid at point 5 mentioned above. Then the refrigerant vapor is completely cooled down in the condenser and gets all liquefied. Then the liquid goes into the pump to complete the cycle. Since the refrigerant must be completely in vapor state at the inlet of the expander and be in liquid state at the inlet of pump, two monitoring holes have been set at point 1 and point 4. Besides, the states of points 1–7 and A–C are observed with pressure and temperature sensors for further thermal analysis. When T_5 is higher than T_2, the valve is open and the recuperator would be turned off.

The shape of the saturated vapor curve of the working fluid at evaporation pressure in T–s diagram dominates the performance of the WHR system. The working fluids are classified into drying, wetting and isentropic types [31,32] according to the different slopes (dT/ds) of the saturated vapor curve in T–s diagram, as shown in Fig. 2. Typical wet fluids are water, ammonia and R143a, whose saturated vapor curve slope is negative ($dT/ds < 0$). Thus, the expansion process may end in the two-phase region if superheating at the inlet of expander is insufficient. And this may cause the damage of expander blades [33], which will greatly reduce the efficiency of the system. Typical dry fluids are R123, R245fa and benzene, which possess a positive slope in the saturated vapor line ($dT/ds > 0$). Their expansion processes end in the superheated vapor region. R11 and R134a are examples of the isentropic fluids. Their saturated vapor line is almost vertical ($dT/ds \approx 0$) in most of the temperature range. The properties of formal organic fluids whose decomposition temperature is beyond 600 K are shown in Table 2. Stine et al. [34] pointed out that, the cycle efficiencies using dry fluids are as high as that of isentropic fluids with IHE used in the cycle, which has been proved by Saleh et al. [31]. So we should focus on how thermodynamic performance is and a typical drying fluid, R123 [35,36], is proposed to be studied in this WHR system.

The thermoelectric materials used in this TEG system are tellurium, antimony, germanium and silver (TAGS) as p-type and PbTe as n-type [28,37], and previous studies on the materials proposed in this paper could achieve good performance as we recapture waste heat from engine exhaust gases whose temperature ranges from 570 K to 870 K. A TEG module consists of 71 thermocouples or 142 thermo-elements. When TEG works in practice, usually a specific array structure of TEG modules is proposed and installed on the exhaust tube. Hence we can acquire the target output voltage and the power by placing the modules in series or in parallel. TEG configuration can be simplified to the mode of M × N as described in our previous work [30], which means M rows and N columns and the modules in the same column are in parallel while columns are in series.

3. Analyses of thermodynamic processes

Fig. 3a and b illustrates the thermodynamic processes under subcritical and supercritical conditions separately. To describe the thermodynamic processes, R123 is selected, while other candidate refrigerants show similar trends. The theoretical cycle consists of the following processes:

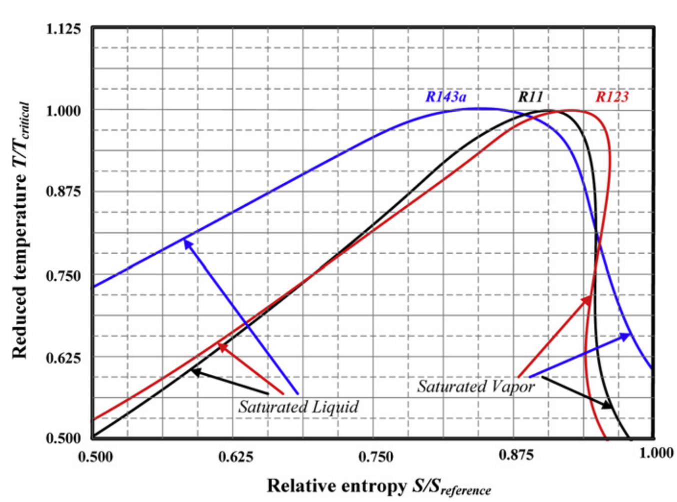


Fig. 2. T–s diagrams for R143a, R11 and R123.

Table 2
Properties of working fluids.

Working fluid	R143a	R11	R123
Molecular weight (g/mol)	84.4	137.37	152.9
$T_{decomposition}$ (K)	650	625	600
p_{CRIT} (MPa)	3.761	4.408	3.662
T_{CRIT} (K)	345.86	471.11	456.8

Under both subcritical and supercritical conditions (p_{evap} = 3 MPa/5.5 MPa) 1–2: Expansion (expander) 2–3: Isobaric internal heat exchange (recuperator) 3–4: Isobaric heat rejection (condenser) 4–5: Compression (boost pump) 5–6–7–1: Isobaric heat supply (recuperator, precooler, evaporator).

4. Thermal modeling of the comprehensive system

In the study, all the calculations and evaluations of the ideal cycles are based on optimized mass flow rate of working fluids where gains maximum net output power. To make the analyses simple and clear, some assumptions are made:

(1) Flow directions of working fluids in recuperator and evaporator are countercurrent, while in TEG are parallel current, and leakage of heat in recuperator and condenser is ignored;
(2) The internal resistances in heat exchangers are negligible and the condensation and evaporation pressures of working fluids keep constant respectively;
(3) Cost analysis for this bottoming system is not included in this paper.

4.1. First law model

In general, a thermodynamic model of thermoelectric generator is established based on the Seebeck, Peltier and Joule effects [38]. The computational model to analyze the TEG was developed using discretization method, as shown in Fig. 4, and was calculated by MATLAB/SIMULINK. The TEG system uses p-type tellurium, antimony, germanium and silver (TAGS)/n-type PbTe materials. The top side of p-type and n-type materials means hot source and the bottom means heat sink. Analysis of heat transfer from the hot junction to the cold leads to the following set of modified energy balance formulas:

For a thermoelectric couple, the heat is absorbed from hot source q_h and released to the cold side q_c,

$$q_h = (T_{exh} - T_h)/r_h = \alpha_{NP} T_h I_o + K_o(T_h - T_c) - 0.5 I_o^2 R_o \quad (1)$$

$$q_c = (T_c - T_{orc})/r_c = \alpha_{NP} T_c I_o + K_o(T_h - T_c) + 0.5 I_o^2 R_o \quad (2)$$

where, T is temperature, q means flow rate of heat in a thermoelectric couple, r is the thermal resistance between fluid and thermo-elements, and α_{NP} dominates the Seebeck coefficient. K_o is thermal conduction, I_o is the current flow and R_o is the electrical resistance in this couple. While the subscript h, c, exh and orc of variables represent the hot junction, the cold junction, the exhaust gas and the working fluid respectively.

In thermo-electric generator, a discretization method is applied. It is supposed that the modules located in the same row have same performance with the same temperature distribution, thus the enthalpy of the exhaust and working fluid in column i ($1 \le i \le 7$) can be expressed as follows,

$$h_{exh}(i+1) = h_{exh}(i) - 71 \times M \times q_h(i)/m_{exh} \quad (3)$$

$$h_{orc}(i+1) = h_{orc}(i) + 71 \times M \times q_c(i)/m_{orc} \quad (4)$$

where, $h_{exh}(1) = h_A$, $h_{exh}(N+1) = h_B$, $h_{orc}(1) = h_7$, $h_{orc}(N+1) = h_8$.

TEG system produces output power, w_{TEG},

$$w_{TEG} = I_{TEG}^2 R_{load} \quad (5)$$

where, the gross electric current, $I_{TEG} = V_{TEG}/(R_{TEG} + R_{load})$. The thermal efficiency of the generator is maximized by choosing an optimum ratio of load resistance to internal resistance [39]. Thus, $R_{load} = R_{TEG}$ is proposed in this paper.

Expansion work output by the expander is,

$$w_T = m_{orc}(h_2 - h_1)\eta_m \quad (6)$$

Pump work input is,

$$w_P = m_{orc}(h_5 - h_4) \quad (7)$$

Thus, net output power is,

$$w_{net} = w_{TEG} + w_T - w_P \quad (8)$$

Indicated thermal efficiency,

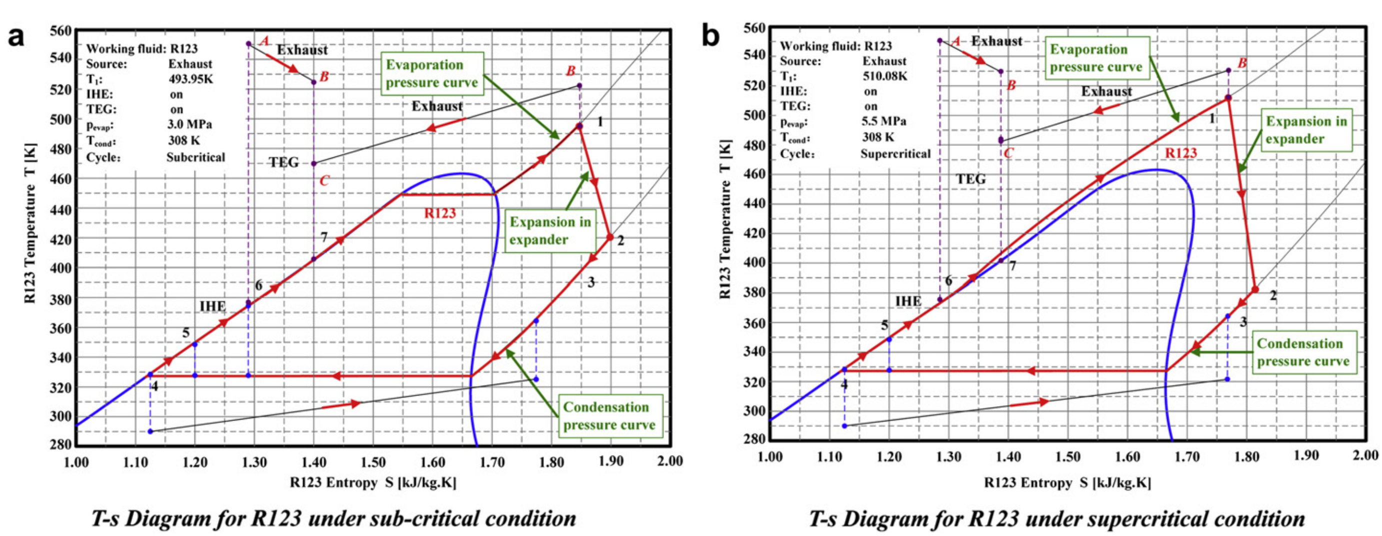
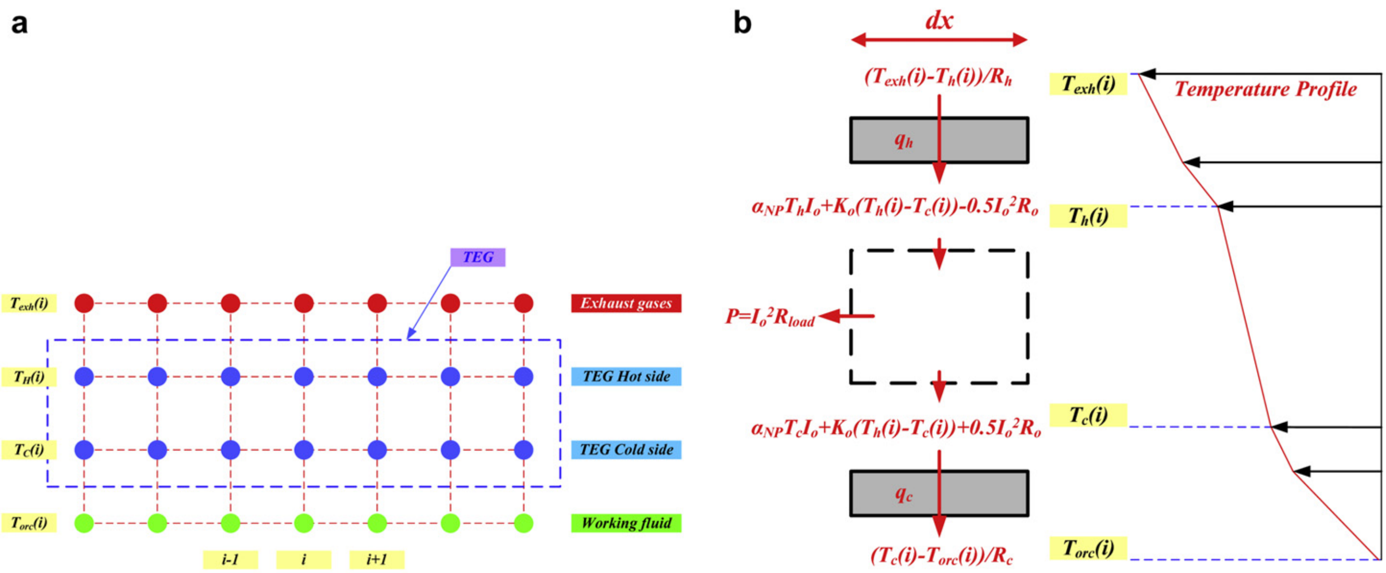


Fig. 3. Thermodynamics analysis of the WHR system.



Discretisation method for heat transfer analysis in TEG *Scheme of heat transfer model*

Fig. 4. Calculation method for TEG.

$$\eta = 3600 \times \left(w_{\text{Engine}} + w_{\text{TEG}} + w_{\text{T}} - w_{\text{P}}\right) / (B \times H_{\text{u}}) \tag{9}$$

where, p is the pressure of working fluid, $\eta_u = 0.8$ is the isentropic efficiency of the expander, $\eta_m = 0.99$ is expander mechanical efficiency, η_{IHE} is the efficiency of the recuperator, $\eta_p = 0.8$ is the isentropic efficiency of the pump, H_u is the low heat value (kJ/kg), $H_u \approx 44100$ kJ/kg, and B is fuel consumption (kg/h).

4.2. Second law model

Exergy loss in evaporator, turbine, IHE, condenser, pump and TEG, respectively:

$$E_{\text{TEG}} = m_{\text{orc}}(e_6 - e_7) + m_{\text{exh}}(e_{\text{A}} - e_{\text{B}}) - w_{\text{TEG}} \tag{10}$$

$$E_{\text{evap}} = m_{\text{orc}}(e_7 - e_1) + m_{\text{exh}}(e_{\text{B}} - e_{\text{C}}) \tag{11}$$

$$E_{\text{turbine}} = m_{\text{orc}}(e_1 - e_2) - w_{\text{T}} \tag{12}$$

$$E_{\text{IHE}} = m_{\text{orc}}(e_2 - e_3 + e_5 - e_6) \tag{13}$$

$$E_{\text{cond}} = m_{\text{orc}}(e_3 - e_4) \tag{14}$$

$$E_{\text{pump}} = m_{\text{orc}}(e_4 - e_5) + w_{\text{P}} \tag{15}$$

Total exergy loss of the system is defined as follows:

$$\begin{aligned} E_{\text{ORC}} &= E_{\text{TEG}} + E_{\text{evap}} + E_{\text{turbine}} + E_{\text{IHE}} + E_{\text{cond}} + E_{\text{pump}} + w_{\text{net}} \\ &= m_{\text{exh}}(e_{\text{A}} - e_{\text{C}}) \end{aligned} \tag{16}$$

where, e is the exergy of working fluid, E means the exergy loss. The reference temperature and pressure are 298 K and 101 kPa respectively, in the exergy analysis.

4.3. Validation of the model

Actually, error analysis should be discussed comparing the calculation results with the test results, because pressure drop, temperature variation and irreversibility in each component are ignored in our calculation model. Thus, error analysis is necessary and important to make sure that the model is accurate. The calculation model of ORC devices is established based on material and energy balance which is widely validated, and has sufficient accuracy [40–42]. So the TEG model is the core part to be validated, and error analysis of TEG is necessary. Since this work is the initial numerical study of the TEG-ORC system, the emphasis is put on the prediction of the overall system performance, and the presented model has been validated by the previous works of other researchers.

The above thermoelectric numerical calculated solution is validated by similar researches and compared with the theoretical values from Angrist [39] and the results from Hussain et al. [43]. They use a thermoelectric device to convert part of the exhaust energy into electricity based on a gas-electric hybrid vehicle. In this comparison, a single thermoelectric couple with a surface area of 224 mm^2 was presented and the desired hot and cold side temperatures were achieved by controlling mass flow rate and temperature of working fluids. These results are in good agreement, as shown in Fig. 5. Indeed, the biggest relative error is less than 1.81%. Moreover, simulations do not show problems such as chattering or numerical oscillations typically observed in discretized models. Further development must rely on extensive engine testing. However there is no experimental data for TEG-ORC system available yet. It is of great necessity and guiding value for the experiment, and experimental data of the whole system compared with calculation value would be published in the following research.

5. Results and analysis

Compared with solar energy and geothermal resource, the heat capacity of exhaust gases is limited, thus the net system output is much more important and attractive than the output per unit of mass flow rate of R123, and system weight and size must be controlled to reduce load and save space when WHR system is applied to vehicle engines. Thus the net output power of this system w_{net} and volumetric expansion ratio v_2/v_1, which determines system performance and the size of turbine expander, were chosen as the two objective functions. In this section, the first law

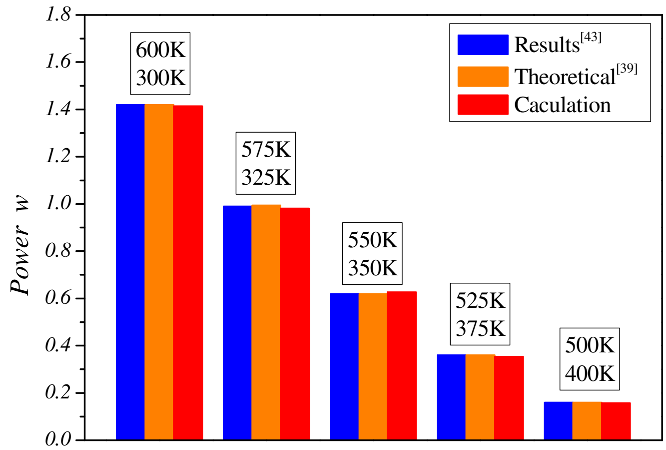


Fig. 5. Comparison of calculated output power vs. results in Ref. [21] and theoretical output in Ref. [20].

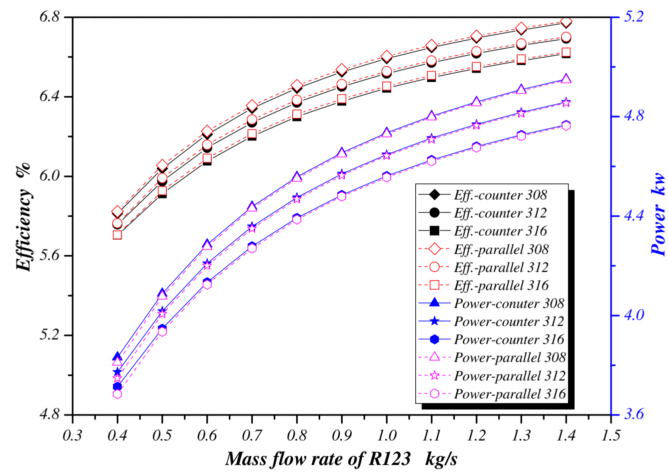


Fig. 6. Comparison of the parallel and counter flow of TEG configuration on efficiency (%) and net output power (kW).

analysis was used to evaluate the system performance by investigating the relative TEG flow direction, TEG scale, highest temperature, condensation temperature, evaporator pressure, and efficiency of internal heat exchanger (IHE). A second law performance has also been studied in target working areas.

The temperature, pressure and mass flow rate of exhaust gases coming from the engine test results are used for the calculation model, and then the performance of the TEG-ORC system has been predicted and analyzed under different working conditions. In the following calculations the boost pump efficiency is set to 0.80 and the turbine expander efficiency is 0.80. The temperature of the exhaust coming out from the engine (point A) is 792 K, and at the evaporator outlet (point C) the temperature is controlled to keep about 500 K through heat exchangers design and the mass flow rate of exhaust gas m_{exh} is 0.2752 kg/s. Based on the above basic equations and initial parameters, the performance of the system is evaluated and the results are discussed in this section. All the parameters used in the calculation come from NIST REFPEOP 7.0 [44], which is with sufficient accuracy.

5.1. First law analysis

5.1.1. Parallel vs. counter flow of TEG

Engine exhaust gas with high temperature of 792 K and mass flow rate $m_{exh} = 0.2752$ kg/s is used as the hot source of the thermoelectric generator, while the cold R123 with the temperature of 308 K, 312 K and 316 K is proposed as the heat sink. The mass flow rate of R123 m_{orc} varies from 0.4 to 1.4 kg/s while the pressure is kept at 4.0 MPa. Thus the performance of the TEG is predicted under both parallel and counter flows.

Fig. 6 shows the TEG performance prediction of the proposed mass flow rate of R123. When the inlet temperature of cold R123 is kept at 308 K, the electricity power generation is initially 3.8 kW at 0.4 kg/s and increases to 4.95 kW at 1.4 kg/s, while the efficiency varies from 5.8 to 6.77% under parallel flow configuration. And as the inlet temperature of R123 gets higher, the output power becomes much lower. That's because larger mass flow rate and lower inlet temperature of R123 result in a lower mean temperature of cold sink. Consequently, the mean temperature difference maintains lager, and thereby a good performance can be achieved by TEG.

When it comes to counter flow, the power and efficiency indicate the similar trend varies with m_{orc} compared to parallel flow. But the efficiency is a little lower than that of parallel flow, while the power is a little higher. Since the generation power takes a small portion of the total power output of TEG-ORC system, parallel configuration with higher efficiency will be proposed in the later sections.

5.1.2. TEG configuration

Through this investigation, the exhaust gases after turbocharger are inducted into the bottoming cycle, and the irreversibility associated with heat transfer and compression processes are neglected. In this calculation, exhaust temperature is controlled and cooled down to about 500 K. The condensation temperature is set at 308 K. Evaporator pressures 3.0 MPa (subcritical) and 5.5 MPa (supercritical) are compared in this section when different scales of TEG were proposed. In this temperature range, the superheat stroke when expansion processes end is considered.

With the scale of TEG becomes larger, more heat is inducted into TEG, and then exhaust temperature at point B (T_B) gets lower before entering evaporator. Lower temperature at inlet of the evaporator can prevent the organic fluid from resolving.

Fig. 7 illustrates the power distribution of turbine expander, TEG, boost-pump as well as net power, and shows the indicated engine efficiency. Under subcritical conditions ($p_{evap} = 3$ MPa), expander output power is kept at about 20.43 kW and pump power about 0.94 kW, while TEG output w_{TEG} increases from 0.42 kW to 3.98 kW as T_B gets lower. Thus the net output power rises from 19.9 kW to 23.5 kW. For supercritical condition ($p_{evap} = 5.5$ MPa), expander output power and pump power are approximately 22.9 kW and 2.0 kW individually, the TEG output w_{TEG} increases from 0.42 kW to 3.9 kW, and thereby net power output ranges from 21.3 kW to 24.7 kW.

The results show that, the exhaust temperature at the inlet of expander reduces when sufficient TEG modules are applied in this system. When T_B is kept at about 600 K, R123 resolving can be prevented, since the decomposition temperature of R123 is about 600 K, thereby the engine with WHR system would work fluently and safely under most conditions. Thus TEG in the mode of 20 × 7 is proposed in the latter sections.

5.1.3. The highest cycle temperature

In this section, the effect of the maximum temperature of bottoming cycle will be scanned, in order to make it clear how the overheating effects system performances, especially for the net output power and expansion ratio required in turbine expander design. Engine exhaust gas is cooled to 600 K and mass flow rate of

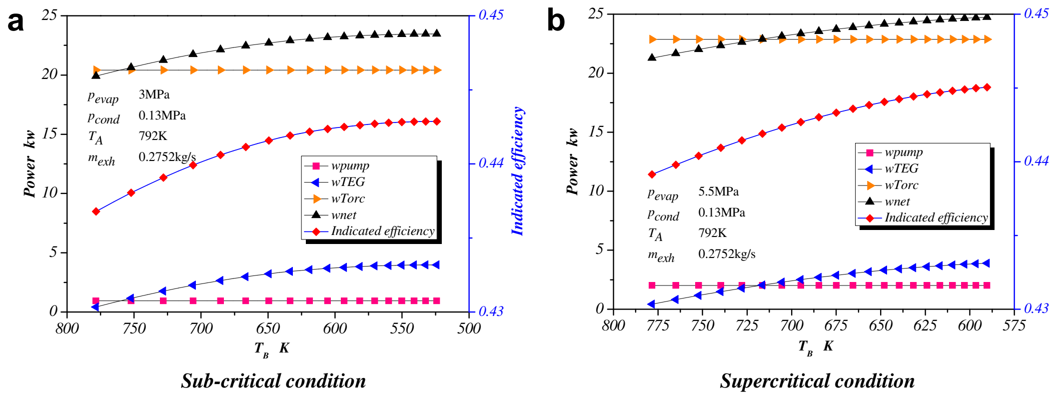


Fig. 7. Variations of the power with different *M*TEG.

R123 $m_{exh} = 0.2752$ kg/s is taken into consideration. In this section, an ORC only system without TEG and IHE is proposed to study the T_{max} effects on system performance. Through controlling mass flow rate of R123, T_{max} varies from 420 K to 590 K. The condensation temperature is set at 308 K.

When T_{max} is not sufficient under high evaporator pressure, the temperature of exhaust from turbine T_2 may be lower than the temperature of pump outlet T_5, thus IHE should be turned off to avoid energy pollution. Besides, IHE is necessary for making full use of exhaust energy and raising power output referred to literature [45,46]. An appropriate highest temperature of the system should be chosen to achieve high performance and thermal efficiency when a WHR bottoming system is designed.

Fig. 8a illustrates the output power per unit of mass flow rate of R123 under different evaporator pressures and also shows total net output variations. The output power per unit of mass flow rate of R123 keeps always rising as the highest temperature T_{max} increases. Thus the ability of R123 for converting heat to power is enhanced. However, the quantity of exhaust heat is limited, which is different from solar energy and geothermal resources. So the net output power of the overall system is considered more important in the area of engine exhaust waste heat recovery and the net output power of the overall system w_{net} is adopted to evaluate WHR system performances in this paper.

As shown in Fig. 8a, under subcritical condition, w_{net} increases firstly and then decreases latter as the highest temperature increases. w_{net} reaches maximum power of 5.03 kW when T_{max} equals to 465 K. Under supercritical conditions, the value of w_{net} keeps increasing, while the increasing trend keeps gentle at the later part. Through the increasing trend we conclude that when T_{max} becomes high enough, output power significantly increases, but if the temperature goes too high, the cost of evaporator will rise. Consequently, T_{max} should be selected properly according to system configuration and further economic analysis.

Fig. 8b illustrates the expansion ratio required by turbine expander. When T_{max} rises, the expansion ratio gets lower under specific evaporator pressure. The calculation results indicate that the working fluid holds enhanced ability of converting heat to power as the highest temperature increases. The cost of evaporator will rise when the size and weight of turbine expander decreases.

5.1.4. The efficiency of IHE

Exhaust gases and working fluid maintain a great temperature difference when TEG is applied to WHR bottoming system, consequently the temperature range of the hot source can be extended for engine security. Two cases, ORC system only to recover heat from exhaust with the temperature of 600 K and TEG-ORC system with the temperature of 792 K, have been considered in this

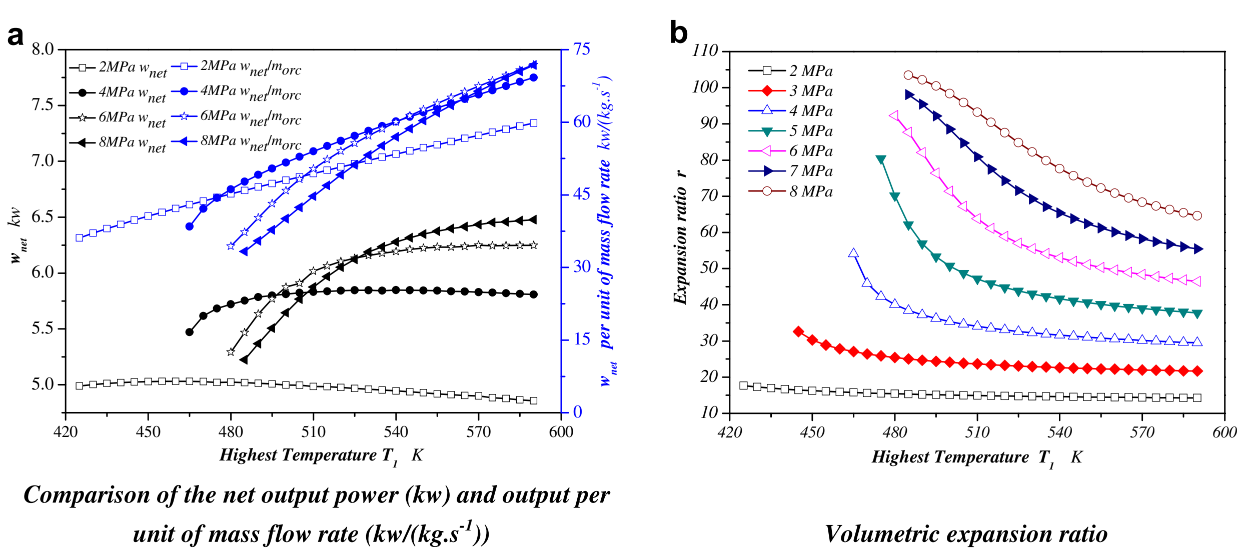


Fig. 8. Performance variation of ORC only system utilizing exhausts with temperature of 600 K.

section. The two cases guarantee that the system can work in a steady and safe state within an acceptable temperature range.

Fig. 9a shows the maximum net output power under various evaporator pressures when an optimized mass flow rate of R123 is proposed. The results indicate that system performance is enhanced with the IHE proposed in this system. For example, when evaporator pressure is set at 5.5 MPa, the ORC only system output is 6.17 kW; and if the IHE is proposed in the system configuration, the output can increase to 8.59 kW when the efficiency of IHE is set at 0.8. The power generated by TEG-ORC system without IHE is 20.04 kW. When IHE is inducted into this system, performance improvement can be achieved. Total output of 26.15 kW can be obtained with the efficiency if IHE is 0.8.

Besides, the output increases significantly as the evaporator pressure increases. When IHE is proposed in TEG-ORC with the efficiency of 0.8, the overall net output power can reach as high as 26.15 kW under the evaporation pressure of 5.5 MPa. When the pressure goes up beyond critical pressure p_{CRIT}, the increasing trend becomes gentle. And the output power in ORC and TEG-ORC systems can be 8.51 kW and 26.15 kW respectively when evaporation pressure is 5.5 MPa and IHE efficiency is 0.8. In the latter case more heat is conducted into evaporator and converted into useful work in expanders.

Meanwhile, Fig. 9b shows the volumetric expansion ratio under maximum power condition will rise within a small scale. IHE can greatly reduce the expansion ratio when the evaporation pressure is below 7 MPa.

Results indicate that more power can be regenerated when TEG-ORC is proposed, since TEG-ORC extends the temperature range and then much more waste heat can get into this bottoming system.

5.1.5. The condensation temperature and evaporator pressure

Fig. 10a shows the cases of turbine expander output power. If condensation temperature keeps constant, as evaporator pressure increases from 2 MPa to 8 MPa, the expander power will increase firstly and then decrease gradually; and if the evaporator pressure keeps constant, the power would increase with the decrease of condensation temperature. When $p_{evap} = 5.5$ MPa and $T_{cond} = 303$ K, the expander output power (28.24 kW) is maximum.

Fig. 10b shows variation of the TEG power with different working parameters. The TEG power decreases a little as the condensation temperature increases, and it keeps decreasing and then increasing as the evaporator pressure rises, while it changes a little within 2.2–2.8 kW during 6 MPa–8 MPa.

Fig. 10c indicates the pump power required in this system. Obviously pump power varies a little at different condensation temperatures, and it increases sharply as the evaporator pressure increases.

Fig. 10d shows the variation of the gross power under different pressures. Keeping the condenser pressure constant, the gross power begins increasing with the evaporator pressure increases from 2 MPa, but it decreases when the evaporator pressure is higher than p_{CRIT}; and keeping the evaporator pressure constant, the gross power would increase as the condensation temperature decreases. When $p_{evap} = 5.5$ MPa and $T_{cond} = 303$ K, the power (27.01 kW) is maximum.

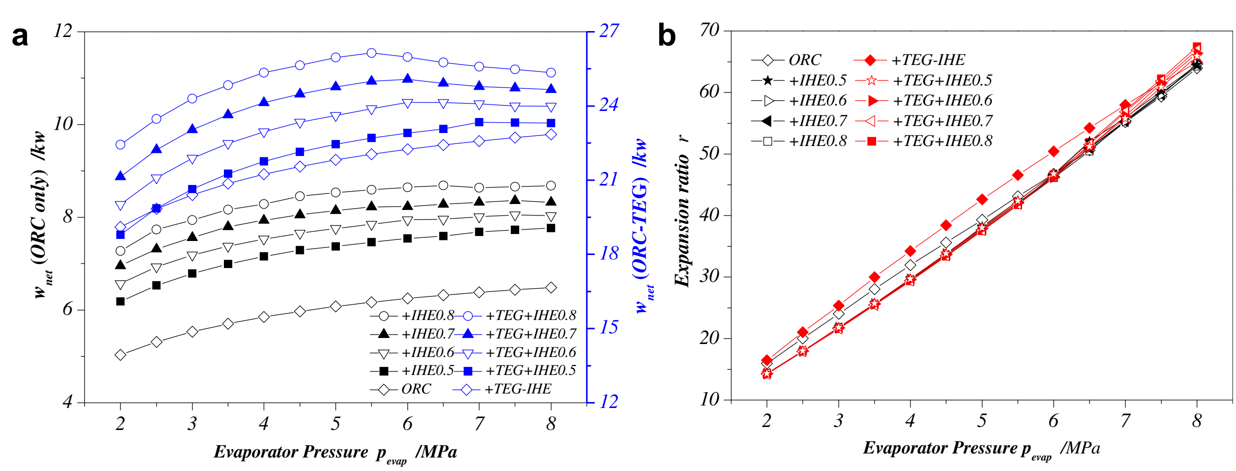
Therefore, evaporator's pressure can be set neither too high nor too low. If too high, the state of the organic working fluid would be unsteady and it requires more devices and costs. If too low, it requires a more efficient condenser and the system sealing would be a challenge. In the tests in practice, the above factors and the system power distribution should be taken into consideration. And the condensation temperature should be as low as possible, which is limited by ambient temperature, thus it can output a considerable amount of power.

We can see that optimal working pressures can be achieved by designing TEG and heat exchangers properly. When the system works under the condition of $p_{evap} = 5.5$ MPa/$T_{cond} = 303$ K, 27.01 kW additional power can be achieved, and the indicated efficiency gets higher from the original 41%–45.71%.

5.2. Second law analysis

In this section, an exergetic analysis of TEG-ORC system with or without IHE has been studied under both subcritical and supercritical conditions. Table 3 shows the thermodynamic parameter distribution in the TEG-ORC. The presented temperature, pressure and other parameters at specific points are predicted based on theoretical analysis when condensation temperature is set at 308 K. And the efficiency of IHE, η_{IHE}, is set at 0.8 if recuperator is proposed. The reference temperature and pressure are 298 K and 101 kPa respectively in the exergy analysis.

As shown in Table 3, main parameters of the four cases are compared. Case I and Case II are working under evaporator pressure of 3.0 MPa without/with IHE, and Case 3 and Case 4 are working under evaporator pressure of 5.5 MPa without/with IHE, respectively. The net output power is 20.86, 22.96, 22.25 and 23.60 kW, and the optimal mass flow rates of R123 are 0.342, 0.423, 0.297 and



Net output power *Volumetric expansion ratio*

Fig. 9. Maximum net output power (kW) and volumetric expansion ratio at different cases.

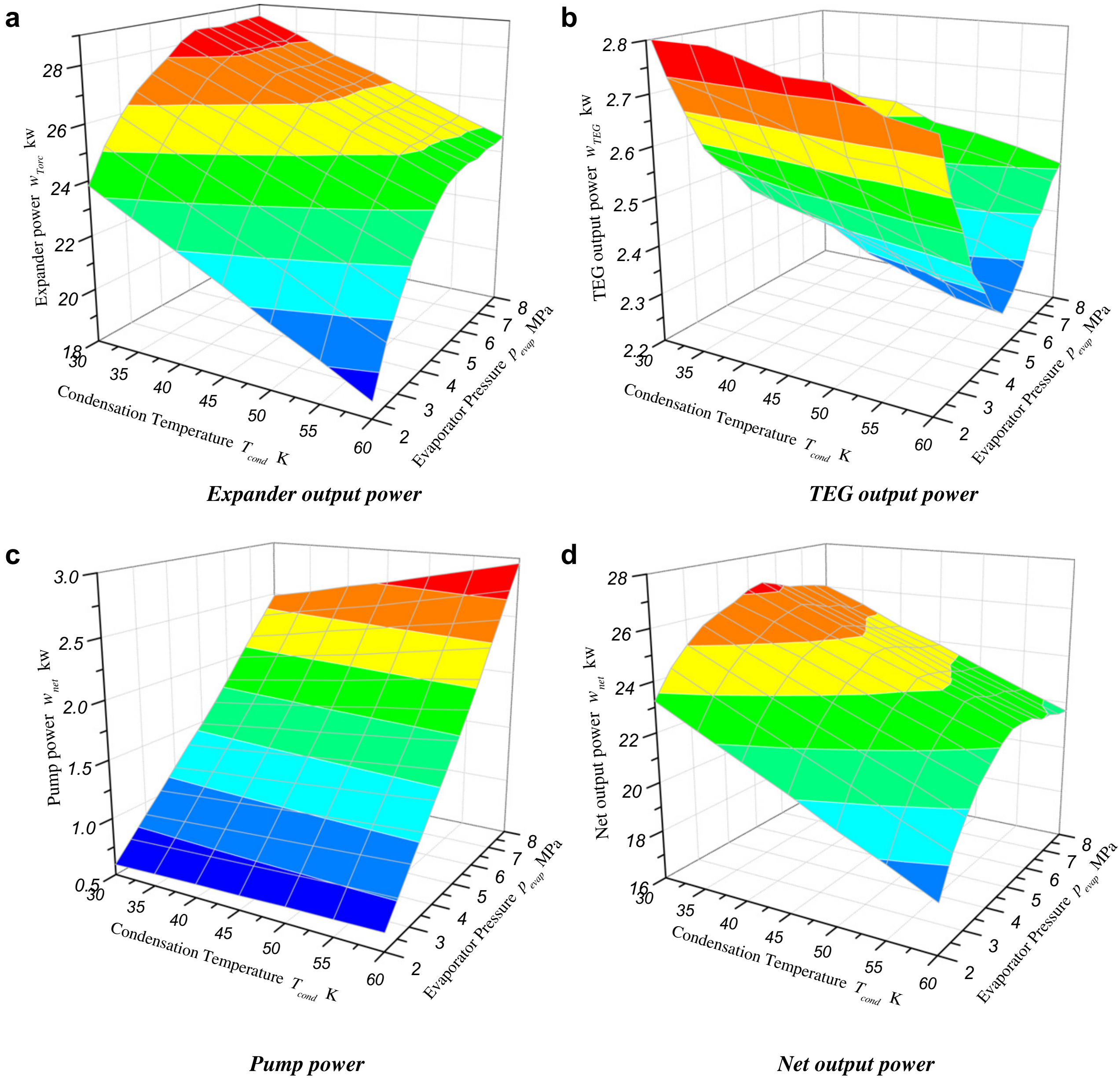


Fig. 10. Performance variation of TEG-ORC system utilizing R123 as working fluid under different evaporator pressures and condensation temperatures.

0.420 kg/s, respectively. The designed ratio of volumetric expansion which is shown in Table 3 greatly affects the weight and size of expander. Lower volumetric expansion ratio helps downsize the expander and requires lower designed intensity.

The relative exergy losses of each component such as evaporator, turbine expander, condenser, pump, TEG and IHE as well as system exergy efficiency are shown in Table 3. The results indicate that the TEG has the largest exergy loss in most cases, while the exergy in pump always keeps lowest, followed successively by the IHE. When TEG-ORC works under subcritical condition, the evaporator possesses a large portion of the total exergy loss, while under supercritical condition the portion becomes much smaller. This finding indicates that once TEG technology gains breakthrough in higher energy conversion efficiency, the system would recapture a substantial amount of power. Table 3 also shows that higher evaporator pressure makes smaller exergy loss in evaporator, but exergy loss gets larger in thermoelectric generator.

The bottoming systems with IHE have larger exergy efficiencies and the efficiencies increase 4.09% under subcritical condition and 2.74% under supercritical condition compared with Cases I and III where IHE is not available, respectively. That's because more energy of turbine exhaust is recovered by IHE, thus, leading to reduced exergy loss in condenser. The IHE is useful especially when the vapor leaving the turbine is superheated. Therefore, a recuperator should be used in bottoming system to improve this TEG-ORC further.

Actually, all the processes in this system are irreversible, and the heat rejected into ambient environment and the friction losses are the main sources of the irreversibility. The actual expansion process is quite different from ideal isentropic process, and the pump and expander frictions also consume part of the converted energy. Thus, the overall thermal efficiency is lower than the ideal cycle. The above analysis demonstrates that, once thermoelectric technology and downsizing design of expander and heat exchangers gain

Table 3
Comparison of proposed TEG-ORC bottoming cycles with/without internal heat exchanger (±IHE) at typical subcritical and supercritical conditions (p_{evap} = 3.0 MPa/5.5 MPa).

Case	I	II	III	IV
Working fluid	R123	R123	R123	R123
System configuration	−IHE	+IHE	−IHE	+IHE
Cycle	Subcritical	Subcritical	Supercritical	Supercritical
Evaporation pressure p_{evap} (MPa)	3.0	3.0	5.5	5.5
Evaporation temperature T_{evap} (K)	444.45	444.45	–	–
Condensation pressure p_{cond} (MPa)	1.3	1.3	1.3	1.3
Condensation temperature T_{cond} (K)	308	308	308	308
T_A (K)	792	792	792	792
T_B (K)	602.3	609.8	608.3	609.7
T_C (K)	500	500	500	500
$T_1 = T_{max}$ (K)	494.0	484.0	554.1	510.1
v_1 (m^3/kg)	0.00671	0.00634	0.00395	0.00281
T_2 (K)	391.7	379.3	428.3	370.4
v_2 (m^3/kg)	0.1603	0.1548	0.1764	0.1509
T_3 (K)	391.7	323.3	428.3	322.6
$T_4 = T_{cond} - 0.5$ (K)	307.5	307.5	307.5	307.5
T_5 (K)	309.4	309.4	310.7	310.7
T_6 (K)	309.4	349.7	310.7	345.3
T_7 (K)	454.0	454.0	481.2	466.4
Expansion ratio $r = v_2/v_1$	30.0	31.1	50.4	63.7
Expansion work output w_T (kW)	18.3	21.7	20.4	23.3
TEG output power w_{TEG} (kW)	3.59	2.55	3.34	2.44
Pump work input w_P (kW)	1.00	1.23	1.51	2.13
Net output power w_{net} (kW)	20.86	22.96	22.25	23.60
WHR thermal efficiency (%)	43.8	43.8	44.1	45.1
Mass flow rate of R123 m_{orc} (kg/s)	0.342	0.423	0.297	0.420
Exergy loss in evaporator (kJ)	9.02	10.62	1.85	1.72
Relative exergy loss in evaporator (%)	17.59	20.71	3.60	3.36
Exergy loss in expander (kJ)	3.94	5.01	4.12	6.09
Relative exergy loss in expander (%)	7.69	9.77	8.04	11.88
Exergy loss in condenser (kJ)	4.99	2.57	6.71	2.54
Relative exergy loss in condenser (%)	9.73	5.01	13.08	4.94
Exergy loss in pump (kJ)	0.31	0.38	0.39	0.55
Relative exergy loss in pump (%)	0.60	0.74	0.76	1.08
Exergy loss in TEG (kJ)	12.15	8.76	15.96	16.07
Relative exergy loss in TEG (%)	23.70	17.09	31.13	31.35
Exergy loss in IHE (kJ)	–	0.97	–	0.70
Relative exergy loss in IHE (%)	–	1.89	–	1.36
Exergy efficiency	40.69	44.78	43.39	46.03

breakthrough, substantial output power enhancement and fuel economy improvement can be achieved. Increase in output power will also reduce specific emissions, since no extra fuel is consumed while considerable additional power is regenerated.

Recent business and literature investigations indicate that additional cost of the waste heat recovery system may not be significant for heavy-duty diesel engines. Once this TEG-ORC system is applied to the heavy-duty diesel engines, fuel saved from the WHR system would be sufficient to recover the added cost in an acceptable period of time.

It should be pointed out that, the relationship among system size, heat increase, cost and efficiency is quite complicated. First of all, downsizing of the main components in design will play an important role in the TEG-ORC system for vehicle applications. Through literature and business investigations we can see that downsizing of the expander costs most, and efficiency of small expander nowadays is quite low, thus the overall system efficiency can't be very high. The expander seems to be the most potential component to lower the overall system size. AVL Company [2] modified a Garret Turbocharger as the expansion device, which can provide a sufficient efficiency. And BNI Company [47] can design turbo-expanders which are small and efficient enough for vehicle applications. But the cost will be significantly high. Once breakthrough is gained in downsizing design of components, a substantial benefit will be achieved. We believe this technology will be mature in five years. Further experimental data should be analyzed to make it clear how the weight and size of the system influence on the fuel efficiency. Detailed information about our designing would be published in further researches. Secondly, the bottoming cycle will influence the performance of diesel cycle in two ways: First is the backpressure rise. The heat exchangers for exhaust gases will lead a rise in backpressure, which will affect the exhaust process in the diesel cycle, and then lower the combustion efficiency of the engine. However it's really hard and complicated to predicate the effects, many researchers ignored the pressure drop when they analyzed exhaust heat recovery [1,5]. Second is the heat increase in the body of the vehicle. The ORC system will bring a heat increase for the whole vehicle body. Thus, the load of the coolant system gets higher, and it will lead a little lower overall efficiency. Now we are emphasizing on the prediction of the overall system performance, thus the effects of the size, economy and heat increase on the overall efficiency are ignored. The relationship should be discussed with the experimental data in the future work.

6. Summary/Conclusions

TEG-ORC system has many benefits compared with the ORC only system in recovering waste heat of engine exhaust gases. The limitation of fluid resolving due to high temperature exhaust is overcome. An energetic and exergetic calculation is conducted for the TEG-ORC system. The effects of relative TEG flow direction, TEG scale, the highest temperature, condensation temperature, evaporator pressure and efficiency of internal heat exchanger (IHE) on the system performance are investigated, and they should be optimized according to system design. The thermodynamic irreversibility that takes place in evaporator, turbine, IHE, condenser, pump and TEG for the TEG-ORC system is investigated under both subcritical and supercritical conditions. For better understanding of the relationship between energy efficiency and energy consumption, further researches should be done based on rebound effect. Besides, the effects of the system size, economy and heat increase on the overall efficiency should be analyzed with the experimental data in the future work.

The following conclusions are drawn from the parametric and exergetic performance investigation carried out in the current paper:

1) The combined TEG-ORC system operates most effectively at the point p_{evap} = 5.5 MPa and T_{cond} = 303 K, the net output power (27.01 kW) is maximum, and the indicated efficiency gets higher from the original 41% (without WHR system) to 45.71%. Although only a small portion of power output is generated by the TEG, this power could play an important role in practice to operate the accessories such as fans and boost pump in engines. The TEG is also important since it cools down exhaust

temperature where the heat is exchanged with R123 in the evaporator, while produces useful power at the same. And preliminary results suggest that optimal selection in working pressures will be important considerations in future designs.

2) The thermodynamic irreversibility that takes place in the evaporator, turbine, IHE, condenser, pump and TEG for the TEG-ORC system is investigated under both subcritical and supercritical conditions. The TEG-ORC system indicates highest exergy efficiency is about 46.03% when the efficiency of IHE is 0.8. The exergy loss in the TEG is the largest. High evaporator pressure and IHE with high efficiency helps gain higher system performance.

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